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ABSTRACT

The need to reduce the refrigerant charge in the air-conditioning and refrigeration industry, together with the requirement of high efficiency, is forcing to search for alternatives to the classical flooded evaporators. This paper presents an experimental study on the heat transfer performances of the Hybrid Film evaporator, a brand new kind of shell-and-tube evaporator currently under patent pending.
Tests were carried out about the heat transfer performances of a reference flooded evaporator, working with constant liquid level, and of a prototype of Hybrid Film, with the same overall dimensions and the same heat transfer area.
The refrigerant considered is R134a, and the fluid cooled is water. The two evaporators were tested one after the other in the same centrifugal, oil-free chiller, condensed by water. In order to evaluate and compare the heat transfer performances, the temperature approaches and the global heat transfer coefficients were measured while varying the compressor load. Their values are presented as a function of the heat flux, together with an estimation of the refrigerant side heat transfer coefficient. Moreover, the effect of the inlet vapor quality was studied.
Finally, the total amount of R134a needed in the chiller for the two different samples is reported, and the refrigerant saving with the new evaporator is discussed.
The analysis of the data collected showed that the Hybrid Film evaporator is a very efficient alternative to the flooded evaporator, as it allows to reduce the refrigerant charge while maintaining, or slightly improving, the heat transfer performances.

1. INTRODUCTION

The impact of the refrigerant charge is growing more and more due to the high cost of the new synthetic fluids (HFO) and the increasing cost of the HFC refrigerants. In the field of the air conditioning, for the centrifugal systems using flooded evaporators it is useful to find alternative heat exchangers with lower refrigerant charge but the same performance. Namely, it is desired a low temperature approach between the water outlet and the evaporating refrigerant, both for increasing the COP of the machine and for reaching low pressure ratios, typically required for centrifugal compressors.
Nowadays, horizontal falling film or spray evaporators start to be broadly considered and used. In these evaporators the tubes are wet by a thin film of liquid. Basically, heat is transferred by conduction and convection across the liquid film with phase change occurring at the liquid-vapor interface and nucleate boiling in the liquid film. As a result, high heat transfer coefficients are reported in the literature. For instance, Zeng *et al.* (1995) found higher heat transfer coefficients for spray evaporation of ammonia on a single plain tube as compared to pool boiling. Roques (2004) and Habert (2009) reported experimental data for falling film evaporation of R134a and R123 on tubes with plain and enhanced surface: their results show that the heat transfer coefficient is higher than in pool boiling. For the same fluids, Zhao *et al.* (2017) also presented experimental data during falling film evaporation on plain and enhanced tubes. Lin *et al.* (2012) measured an increase of the heat transfer coefficient as compared to pool boiling

when using nozzles to spray the refrigerant on the outside surface of the tubes. Zhao *et al.* (2018) reported measurements of local and bundle average heat transfer coefficient under falling film evaporation of R134a.
Nonetheless, spray or falling film evaporators present some complications, which have made difficult the spread in the market. They require a complex distribution system for the liquid inlet and for the flash gas dissipation, and much care in the alignment of the tubes, in order to ensure uniformity in the feed of the liquid and avoid dry patches on the tubes wall. A review of the studies focused on design and effects of the distribution system and tube layout can be found in Ribatski and Jacobi (2005), Abed *et al.* (2014). Moreover, the performance of the evaporator at partial loads can be problematic, as in general it is difficult to maintain wet all the tubes, especially the ones located in the lowest rows. Depending on geometry, operating conditions and fluid properties, different flow patterns can occur (Wang and Jacobi, 2012). Indeed, the two-phase flow pattern plays a role in the thermal modeling. For example, Zhao *et al.* (2016) developed specific correlations as a function of the flow regime.
Vapor escape from the shell is another critical issue which must be properly considered. In classical falling film evaporators, a big volume of the shell is usually required to avoid liquid flowing to the compressor. Besides, interaction of ascending vapor with the liquid film can affect the heat transfer performance, as investigated by Ji *et al.* (2016).
The new design proposed in this work, the Hybrid Film Evaporator, was developed in order to reduce the refrigerant charge maintaining, or improving, the performances of the flooded evaporator.
It belongs broadly to the family of the falling film evaporators, thus a particular attention was dedicated to the compactness. The aim was not exceeding the dimensions of the flooded evaporator, while accurately avoiding liquid entrainment to the compressor.
The concept and operation of the Hybrid Film evaporator is described in the present paper. An experimental comparison between the performance and refrigerant charge of a prototype of Hybrid Film evaporator and a flooded evaporator, with the same overall dimensions and the same heat transfer area, is then reported and discussed.

2. THE HYBRID FILM PROTOTYPE

2.1 The concept of Hybrid Film Evaporator

The new proposed kind of evaporator belongs, in general, to the family of the falling film evaporators, where the liquid refrigerant (together with the flash gas generated in the expansion valve) flows from the top and progressively wets the tube bank in the vertical direction (Figure 1).

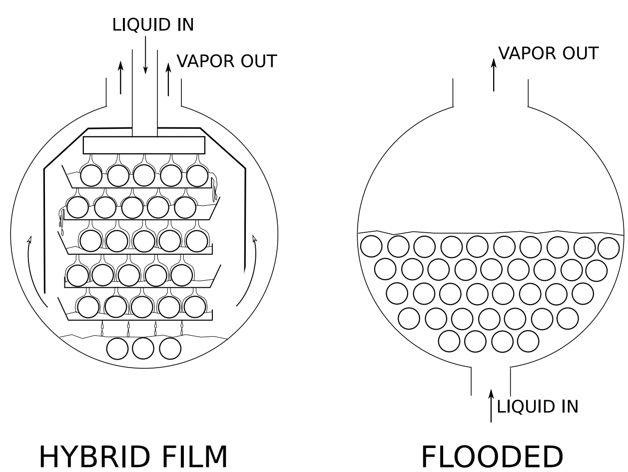

Figure 1: Schematic view of the Hybrid Film and flooded evaporators tested.

The liquid enters inside the inlet nozzle placed at the top, at the middle of the axial direction of the evaporator. Then it passes through a first distribution box, flooding the first tube row. If compared to the solutions presented in the reviewed falling film and spray evaporators, the inlet distribution device is very simple.
In order to maintain an efficient distribution at every condition, the liquid refrigerant is re-distributed at every row of the tube bundle. This aim is obtained by means of special plates placed below every tube row. The plates present two main innovative peculiarities. First, they have special holes that allow the liquid to reach the rows of tubes, and the plates, immediately below. Considering a cross section of the shell normal to the longitudinal axis, each hole is

placed in correspondence with the tube at its bottom, presenting then the same horizontal pitch of the tube bank. In such way, the liquid flows (more or less) in the form of columns, from the upper plate to the lower tube. The diameter of the holes varies from plate to plate: its values were optimized by means of previous studies, both theoretical and experimental, by means of prototypes. The optimization was performed with the purpose to get a continuous feed of the tubes and maximize the heat transfer coefficient. Thus, given the value of the diameter, the number of holes on the axial direction is such to obtain an opportune constant value of the liquid velocity. The Figure 2 shows the bottom tubes invested by the liquid columns from the above plate, visible too.

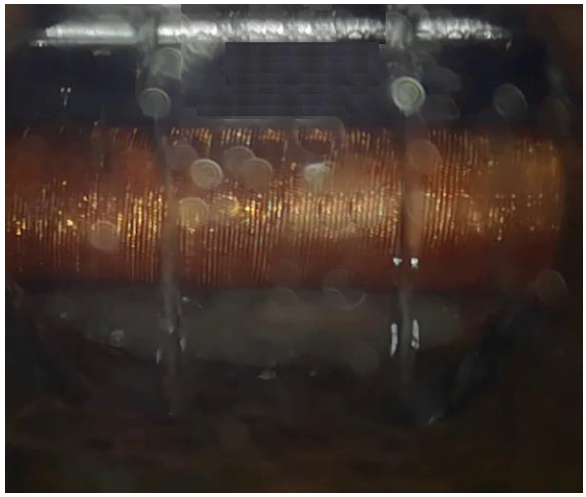

Figure 2: Picture of a tube of the Hybrid Film Evaporator wet by the liquid.

The second characteristic of the plates is that they have vertical longitudinal borders, seemingly like bowls containing the entire tube row. One of the two border is high and it reaches the above plate, literally closing on one side. The other one is shorter, and it forms an opening that allows the vapor to escape and the liquid to fall down inside the bowl immediately below. The kind of the borders is alternate from plate to plate (Figure 1), leaving the liquid to fall down first on the right, then on the left, and so on.
The name “Hybrid Film” comes from the circumstance that in every bowl there is a liquid pool that partially floods the tubes, and a liquid film (more precisely, a set of columns) that wets the tubes from the top.
The expectation is that, varying the load degree, the level of the liquid pool varies consequently. Then, the combined effect of the pool flooding and of the tube wetting allows to always take advantage of the entire tube surface, independently from the load and from the vertical position.
The excess liquid, still to be evaporated, is collected on the bottom of the shell, where a little number of tubes operates as in a typical pool boiling process. The liquid level has to be maintained just above these tubes.
The entire vapor generated by the process, in addition to the flash vapor initially entered, is forced to pass circumferentially, by means of a shield in carbon steel covering the tube bank (Figure 1). Its shape is studied first to minimize the pressure drop, and then the saturation temperature loss, of the vapor. Second, to avoid the entrainment of the liquid drops to the compressor, particularly sensitive, being centrifugal. The vapor is finally collected inside an outlet nozzle, placed on the top of the shell and axially near to one of the tubesheets.

2.2 The samples tested

As introduced above, the reference to evaluate both the performance and the refrigerant charge is the classical flooded evaporator. The prototype of Hybrid Film Evaporator was tested and compared with an equivalent flooded evaporator, with a tube layout as in Figure 1.
For both the kinds of evaporators, the length is 1 m, and the shell diameter 0.61 m. The extreme compactness was one of the characteristic imposed for the samples. The tube pass configuration was chosen in order to keep low the value of the water pressure drop, according to the worldwide growing importance of the reduction of the energy consumption due to the pumps.
The tubes used have the internal and external surface finned. They are specifically designed for flooded evaporators, with a special shape of the fins in order to create additional nucleation sites. The model is Gewa-B6H, provided by Wieland, with outside nominal diameter 19.05 mm.

The number of tubes in both the evaporators is similar, thus the heat transfer surface is almost equivalent.
Both the evaporators were tested one after the other, in the same test rig, using the same compressor, condenser, subcooler and expansion valve.
The flooded evaporator was tested maintaining constant the liquid level as in Figure 1, in order to flood all the tubes at every compressor load. It means that the comparison was carried out with a flooded evaporator whose efficiency was at its best, with the contribution of its entire heat transfer surface at every operative condition.

3. EXPERIMENTAL PROCEDURE

3.1 Experimental test rig

The experimental tests were run in the test facility set up at the laboratory of Onda S.p.A., Italy. It consists of a vapor compression system working with R134a as refrigerant. The compressor is a centrifugal oil free type. A schematic of the apparatus is shown in Figure 3.

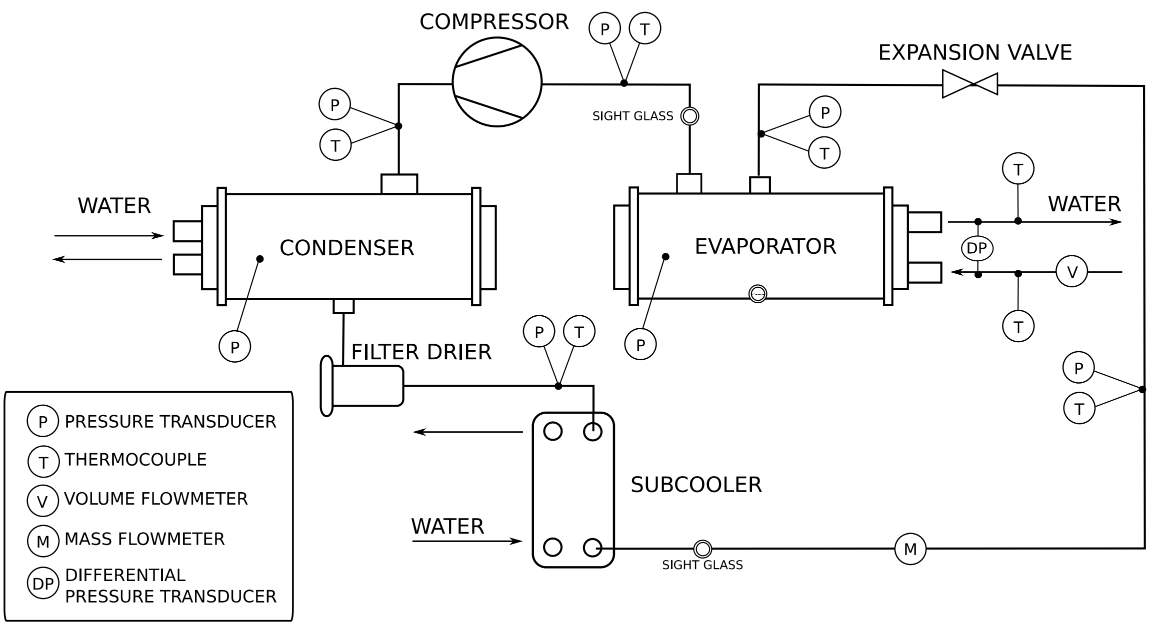

Figure 3: Schematic drawing of the experimental test rig (special case with the Hybrid Film evaporator).

The superheated vapor flows from the compressor to the condenser, where it is fully condensed and then further subcooled in a subsequent heat exchanger. The subcooling level can be independently controlled, varying the cooling water temperature and flow rate. As a result, it is possible to achieve the set vapor quality at the inlet of the test evaporator acting on both refrigerant subcooling and expansion valve.
A step motor electronic expansion valve is used. Its opening is controlled manually by the operator, by varying the voltage supply. A sight glass placed on the evaporator allows to maintain constant the liquid level.
After reducing pressure through the expansion valve, the refrigerant flows to the evaporator and finally to the compressor, again. A sight glass is placed between the evaporator and the compressor to check that there is not entrainment of liquid drops in the vapor flow. In order to make sure, the isoentropic efficiency of the compressor is measured, and compared to the one declared by the compressor manufacturer.
Water is used as secondary fluid to vaporize R134a. The water temperature at the inlet and outlet of the test evaporator is measured by thermocouples inserted in adiabatic sections, where perfect mixing is assured. Type J thermocouples connected to a Kaye ice point instrument as zero reference system are used. Thermocouples were calibrated on site using a primary Pt100 probe and a reference bath to get the accuracy value reported in Table 1. The water volume flow rate and pressure drop through the evaporator are measured with an electromagnetic flow meter and a differential pressure transducer, respectively. Thermocouples and pressure transducers are also installed in different sections of the refrigerant loop to get all the measurements needed for the data reduction. In particular, saturation temperature at the evaporator is obtained by the pressure, which is taken both inside the shell and at the evaporator outlet. A Coriolis effect flow meter measures directly the refrigerant mass flow rate. In this way, the

thermal balances on both water and refrigerant side are compared and checked. The mean deviation is found to be around 3%. The accuracies for sensors and parameters (described in the next section) are listed in Table 1.

Table 1: Accuracy for sensors and parameters (at typical test conditions).

Parameter	**Accuracy**
Temperature	±0.05 K
Water volumetric flow rate	±0.5 %
Refrigerant mass flow rate	±0.5 %
Absolute pressure	±1 kPa
Differential pressure	±200 Pa
Temperature difference	±0.10 K
Temperature approach	±0.12 K
Heat flux	±3 %
Global heat transfer coefficient	±5 %

3.2 Operative conditions and data reduction

In order to understand the behavior at different loads, the compressor capacity was varied by means of an inverter, maintaining constant the evaporator inlet vapor quality. The value was fixed at 0.2, being the evaporator inlet vapor quality of a typical, high efficiency vapor compression system with R134a (condensing temperature 36 °C, subcooling 2 K, evaporating temperature 6 °C).
In a different set of tests, the vapor quality was varied at constant heat flowrate.
The heat flux at the evaporator q_S is given by the water side heat flowrate divided by the reference area A_{ext}, that is the external envelope plain tube surface area at the nominal tube diameter. It ranged from 19 to 31 kW/m².
The temperature approach considered is obtained from:

$$\Delta T_{app} = T_{W,out} - T_{R,sat,out} \tag{1}$$

where $T_{R,sat,out}$ is the saturation refrigerant temperature obtained from the pressure measured at the outlet nozzle. The values of ΔT_{app} ranged from 1.1 to 1.9 K.
The global heat transfer coefficient K is given by:

$$K = q_W/(A_{ext}\, \Delta T_{ml}) \tag{2}$$

where ΔT_{ml} is the logarithmic mean temperature difference between the water and the refrigerant in the shell:

$$\Delta T_{ml} = (T_{W,in} - T_{W,out})/\ln[(T_{W,in} - T_{R,sat,shell}) /(T_{W,out} - T_{R,sat,shell})] \tag{3}$$

The range for K was 6500 – 9500 W/(m² K), and the range for ΔT_{ml} was 2.4 – 3.8 K.
In order to calculate the refrigerant heat transfer coefficient, the authors assume the value of the tubeside heat transfer coefficient by means of a correlation provided by the tube manufacturer. The correlation has been previously verified by means of ad hoc single-phase experimental tests, showing accuracy in the estimation of the experimental heat transfer coefficient. The external heat transfer coefficient α_e is then easily obtained:

$$\alpha_e = 1/[1/K - R_t - d_e/(d_i\, \alpha_i)] \tag{4}$$

where R_t is the conductive thermal resistance of the tube, and d_e and d_i the external and internal nominal tube diameters, respectively. It should be noted, anyway, that the present procedure can be affected by high uncertainty.
All the thermophysical properties of R134a and of the water are calculated by means of the Refprop 9.1 software (Lemmon *et al*., 2013).

4. EXPERIMENTAL RESULTS

4.1 Evaporator performance and refrigerant charge: comparison with the flooded type

At a fixed capacity load degree, the water pressure drop was the same for both the evaporators, as they have almost

the same number of tubes with the same pass configuration. At nominal capacity the pressure drop was around 30 kPa, then inside the limit commonly asked by applications.
A first comparison of the heat transfer performance is shown in Figure 4 and Figure 5. There, the temperature approach ΔT_{app} between water and outlet refrigerant (equation (1)) is plotted as a function of the heat flux.

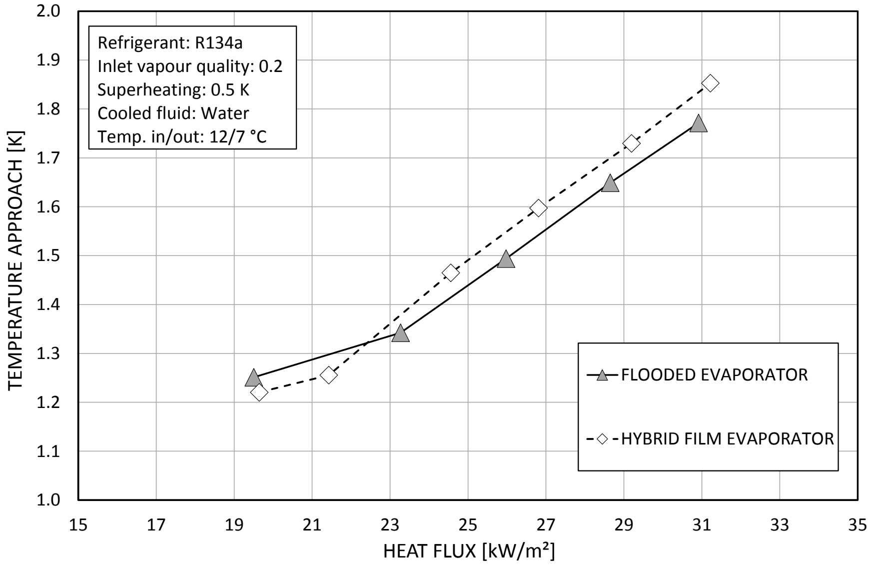

Figure 4: Temperature approach ΔT_{app} between the saturated refrigerant and the water outlet at ΔT_w = const. ($T_{W,in}$=12 °C, $T_{W,out}$=7 °C).

The experimental results in Figure 4 refer to constant water temperature drop, namely 5 K, with the inlet and outlet water temperatures at 12 and 7 °C respectively.
In Figure 5, instead, the water velocity and the water outlet temperature are constant and at the same value for both the evaporators. The water always leaves at 7 °C, and its Reynolds number is around 20000, at a fully developed turbulent flow regime.

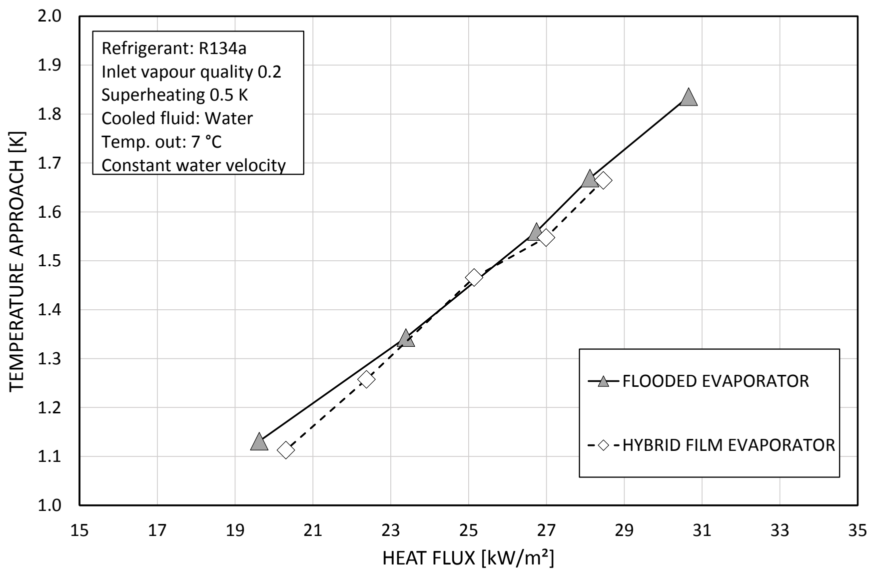

Figure 5: Temperature approach ΔT_{app} between the saturated refrigerant and the water outlet at u_w = const. and $T_{W,out}$ = const. =7 °C.

Figure 6 and Figure 7 report the global heat transfer coefficient (equation (2)) as a function of heat flux, for both evaporators at the same operating conditions of Figure 4 and Figure 5. In Figure 6 the global heat transfer coefficient is strongly dependent on the heat flux. Observing the Figure 7 it results that such trend is due almost exclusively to the increase the internal heat transfer coefficient.

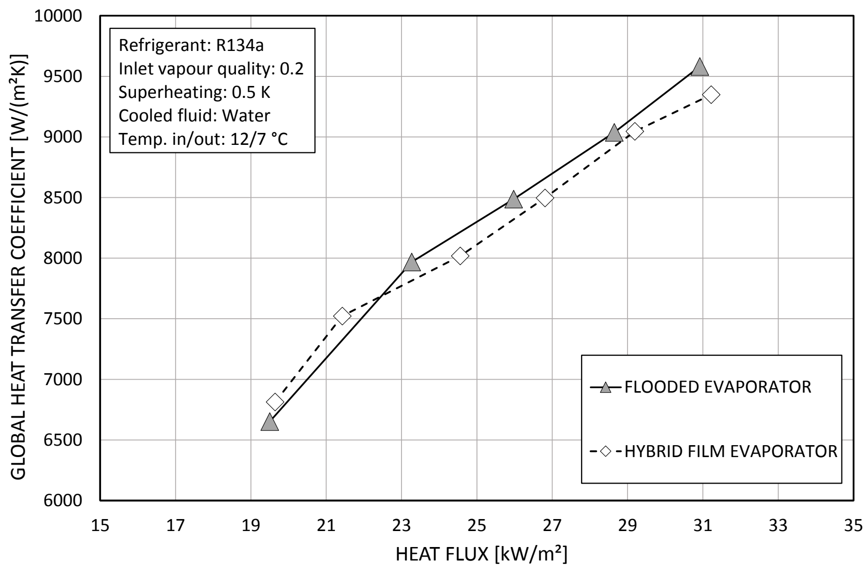

Figure 6: Global heat transfer coefficient at ΔT_w = const. ($T_{W,in}$=12 °C, $T_{W,out}$=7 °C).

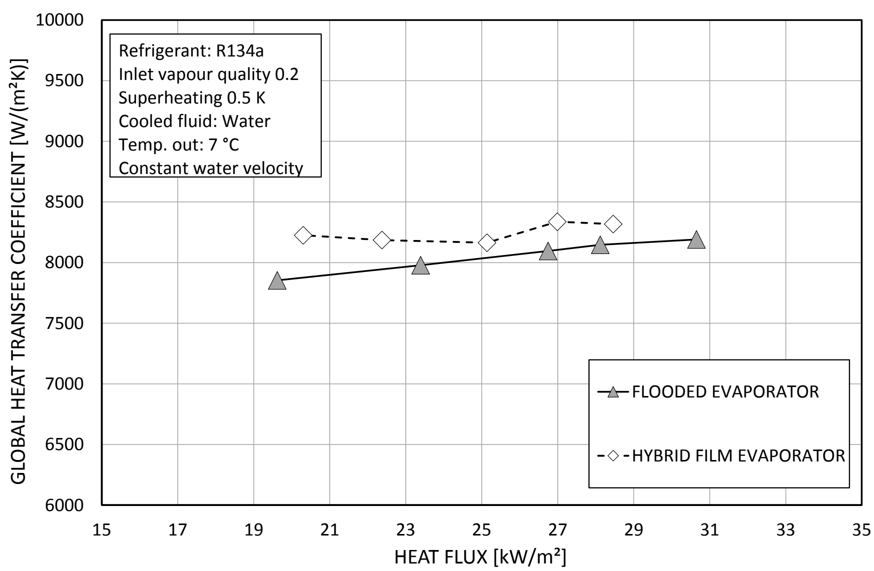

Figure 7: Global heat transfer coefficient at u_w = const. and $T_{W,out}$ = const. =7 °C.

In order to compare the refrigerant charge, the amount of R134a used for the test was accurately weighted for both the flooded and Hybrid Film evaporators. It resulted a total refrigerant charge in the chiller of 112 kg when the flooded evaporator was installed, and of 72 kg when the Hybrid Film evaporator was installed. For the Hybrid Film, then, the amount of refrigerant resulted 36 % less than in the flooded case, with reference to the entire chiller.

4.2 Estimation of the refrigerant heat transfer coefficient

In order to investigate the results of the Figure 7 more accurately, Figure 8 shows the refrigerant heat transfer coefficient estimated as in equation (4). As reported, the uncertainty can be very high, but the internal heat transfer coefficient is almost the same for all the data points reported in Figure 8. Thus, the corresponding uncertainty is the same too.

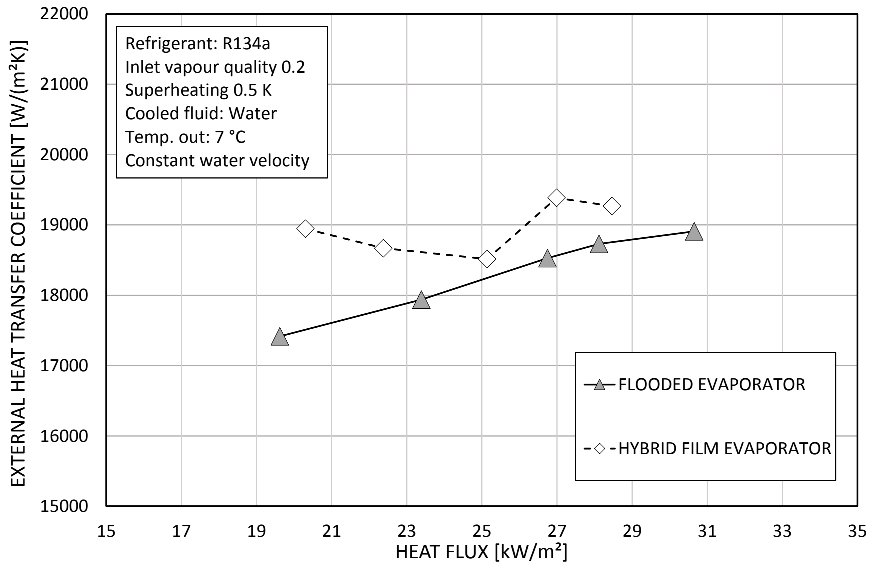

Figure 8: Estimated external heat transfer coefficient at u_w = const. and $T_{W,out}$ = const. =7 °C.

In Figure 8 the flooded evaporator data are well-fitted by an equation like the following:

$$\alpha_e = A\ q_s^B \tag{5}$$

where the value of the slope obtained from the tests is B = 0.2, which is lower than the typical value for plain surfaces (around 0.7, Cooper, 1984), as reported in the literature for tubes with enhanced surface (Cavallini *et al.*, 2005). For the Hybrid Film, instead, the slope is almost null, and the trend is irregular. The heat transfer coefficient does not decrease while increasing heat flux, differently, for instance, from the results by Zhao *et al.* (2018) for falling film evaporation. This behavior will be investigated more accurately in future works.
In addition, the effect of the vapor quality on the heat transfer coefficient was investigated. The Figure 9 reports the values of the heat transfer coefficient at different values of the inlet vapor quality, with constant nominal heat flux.

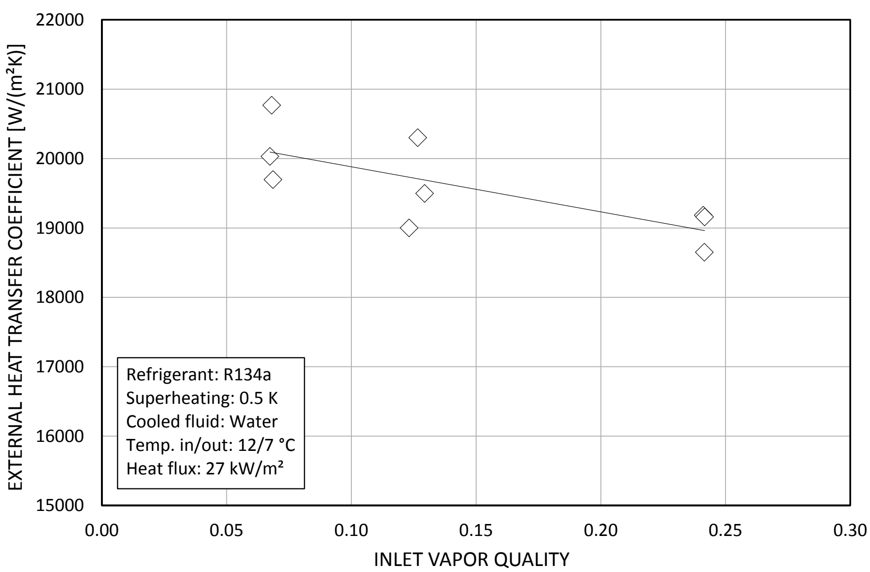

Figure 9: Estimated external heat transfer coefficient at constant heat flux, as a function of the inlet vapor quality.

It results a slight dependence on the inlet vapor quality: the heat transfer coefficient decreases as the vapor quality increases. This is probably due to the distribution process, affected by the growing amount of flash vapor. It should be noted, anyway, that there is also some increase of the refrigerant flow rate with vapor quality, the data being reported at constant heat flow rate.

4.3 Refrigerant pressure drop

The total pressure drop of the refrigerant (from the inlet to the outlet nozzles) is reported in Figure 10. For the Hybrid Film it is significantly lower than for the flooded evaporator. Thus, there is no need to produce a large pressure drop in the distributor: this is a significant difference from spray evaporators, where the nozzle configuration and pressure drop can play a key role on the heat transfer coefficient, as discussed by Zeng *et al.* (1995) and Ribatski and Jacobi (2005). Moreover, a very tiny fraction of the flash vapor is generated inside the heat exchanger.

The low refrigerant pressure drop of the Hybrid Film evaporator can be seen as an advantage, when compared to other solutions for the refrigerant charge reduction. For spray evaporators, and for falling film evaporators with complex distribution devices, this pressure drop is generally high: sometimes its value exceeds 1 bar. If the compressor pressure ratio is low (as in centrifugal chillers), it may represent a problem when sizing the expansion valve.

Finally, the contribution of the vapor collection is low too. In particular, the authors measured a drop in the saturation temperature between the shell and the outlet nozzle ranging from 0.2 to 0.4 K.

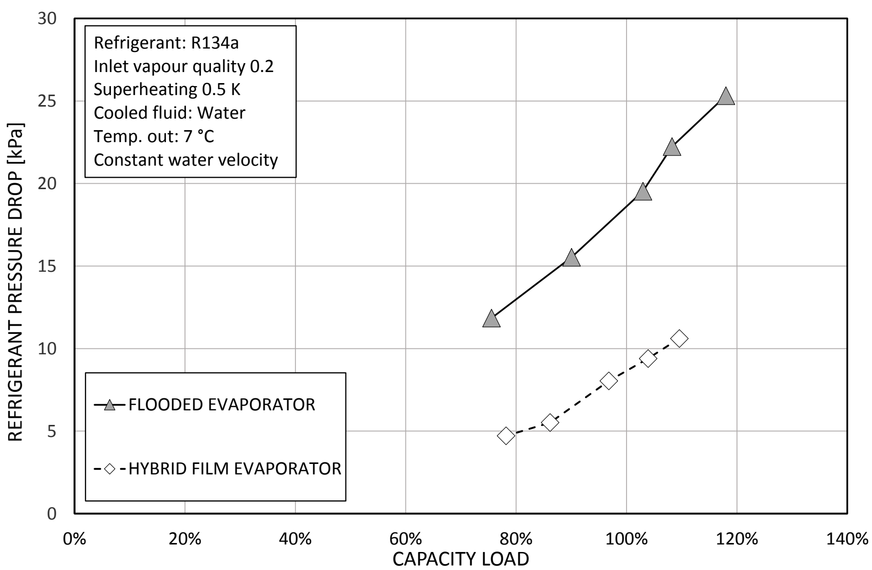

Figure 10: Refrigerant pressure drop as a function of capacity load: 100% load corresponds to 27 kW/m².

6. CONCLUSIONS

A new kind of shell-and-tube heat exchanger, the Hybrid Film evaporator, is presented in this work. It is an evaporator where the liquid flows from the top, and is re-distributed at every row of tubes, allowing to take advantage of the entire heat transfer surface at every operative condition.

The experimental results of comparative tests on a centrifugal oil free chiller with R134a show that:

- the Hybrid Film evaporator reaches, or slightly exceeds, the heat transfer performances of the flooded evaporator;
- at partial loads, the refrigerant heat transfer coefficient of the Hybrid Film evaporator remains almost constant;
- the refrigerant pressure drop in the Hybrid Film evaporator is very low, even less than in the flooded evaporator;
- the refrigerant charge in the Hybrid Film evaporator is considerably less than in the flooded evaporator: around 36 % lower in mass for the entire chiller.

The positive results obtained with R134a suggest similar advantages even for new HFO refrigerants: the authors plan a future study of the Hybrid Film evaporator with R1234ze and R513A.

NOMENCLATURE

q_S	heat flux	[W/m²]
T	temperature	[K]
u	velocity	[m/s]
α	heat transfer coefficient	[W/(m² K)]
ΔT	temperature difference	[K]

Subscript

app	approach
in	inlet
ml	mean logarithmic
out	outlet
R	refrigerant
W	water

REFERENCES

Abed, M. A., Alghoul, M. A., Yazdi, M. H. & Al-Shamani, A. N. (2015). The role of enhancement techniques on heat and mass transfer characteristics of shell and tube spray evaporator: a detailed review. *Applied Thermal Engineering*, 77, 923-940.

Cavallini, A., Del Col, D., Padovan, A. & Censi, G. (2005). Experimental setup for investigation of boiling heat transfer on tube bundles. *Proceedings of IIR International Conference, Thermophysical Properties and Transfer Processes of Refrigerants, Vicenza, Italy*.

Cooper, M. G. (1984). Saturation nucleate pool boiling, a simple correlation. *Proceedings of 1st U.K. National Conference on Heat Transfer, Leeds, U.K.*.

Habert, M. (2009). *Falling film evaporation on a tube bundle with plain and enhanced tubes*. Ph.D. Thesis Ecole Polytecnique Fédéral de Lausanne, Department of Mechanical Engineering.

Jacobi, A. M., Wang, X. (2012). A thermodynamic basis for predicting falling-film mode transitions. *Proceedings of the International Refrigeration and Air Conditioning Conference at Purdue, West Lafayette, IN.*

Ji, W-T., Zhao, C.-Y, Zhang, D.-C., Yoshioka, S., He, Y.L. & Tao, W.-Q. (2016). Effect of vapor flow on the falling film evaporation of R134a outside a horizontal tube bundle. *International Journal of Heat and Mass Transfer*, 92, 1171-1181.

Lemmon, E. W., Huber, M. L., McLinden, M. O. (2013). NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg.

Lin, R.-L., Chang T.-B. & Liang C.-C. (2012). Effects of spray incident angle on heat transfer performance of rhombous-pitchshell-and-tube interior spray evaporator. *Journal of Mechanical Science and Technology*, 26, 681-688.

Ribatski, G. & Jacobi, A. M. (2005). Falling-film evaporation on a horizontal tubes a critical review. *International Journal of Refrigeration*, 28, 635-653.

Roques, J.-F. (2004). *Falling film evaporation on a single tube and a tube bundle*. Ph.D. Thesis Ecole Polytecnique Fédéral de Lausanne, Department of Mechanical Engineering.

Zeng, X., Chyu, M. C., Ayub, Z. H. (1995). Evaporation heat transfer performance of nozzle-sprayed ammonia on a horizontal tube. *ASHRAE Transactions*, 103, 136-149.

Zhao, C.-Y., Ji, W.-T., Jin, P-H. & Tao, W-Q (2016). Heat transfer correlation of the falling film evaporation on a single horizontal smooth tube. *Applied Thermal Engineering*, 103, 177-186.

Zhao, C.-Y., Ji, W.-T., Jin, P-H., Zhong, Y.-J. & Tao, W-Q (2018). Experimental study of the local and average falling film evaporation coefficients in a horizontal enhanced tube bundle using R134a. *Applied Thermal Engineering*, 129, 502-511.

Zhao, C.-Y., Jin, P-H. Ji, W.-T. & Tao, W-Q (2017). Experimental investigations of R134a and R123 falling film evaporation on enhanced horizontal tubes. *International Journal of Refrigeration*, 75, 190-203.